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STRESS-LIFE RELATION OF THE ROLLING-CONTACT FATIGUE SPIN RIG

By Robert H. Butler and Thomas L. Carter

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

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SUMMARY

The rolling-contact fatigue spin rig was used to test balls of a bearing steel at stresses higher than those produced in bearings. Groups of 9/16-inch SAE 52100 grade 1 steel balls lubricated with a mineral oil were tested at 600,000-, 675,000-, and 750,000-psi maximum Hertz stress. The balls were tested at room temperature on fixed tracks that were randomly oriented to the fiber direction of the ball. Plain cylinders of commercial- and vacuum-melt AISI M-1 and commercial-melt AISI M-50 (MV-1) with a bore size ranging from 3.250 to 3.550 inches were used as race specimens.

Stress-life exponents of 10.4 for the SAE 52100 balls and 9.7 for the MV-1 cylinders were obtained. These values are in reasonably good agreement with the values (9 to 10) accepted by the industry. Anomalous results were obtained from a group of AISI M-1 cylinders. Fatigue spalls of subsurface origin similar to those found in complete bearings were obtained. This, plus the agreement in stress-life exponent, establishes the validity of the test for materials comparison and basic investigation of the many factors involved in rolling-contact fatigue.

INTRODUCTION

The application of rolling-contact bearings in aircraft gas-turbine engines has created a complex problem that is commonly referred to as the high-temperature bearing problem (refs. 1 to 4). Rolling-contact bearings, if properly designed, installed, and lubricated, do not wear out by abrasion but fail by fatigue of the races and rolling elements. In addition to the requirement that the material have a satisfactory rolling-contact fatigue strength, other criteria are considered in selecting a suitable bearing steel for high-temperature applications. Of these criteria, hot hardness and dimensional stability are the most important (ref. 3).

The steel used in precision rolling-contact bearings at present is SAE 52100 (a chromium-alloy steel). This steel is temperature-limited in that, above 400° F, its hardness drops below the minimum of Rockwell C-58 accepted by the industry for highly loaded ball bearings (ref. 3). Low hardness increases the possibility of mechanical damage to the bearing and reduces its fatigue strength.

The requirement that a bearing steel must retain a hardness above Rockwell C-58 throughout its operating temperature range (650° to 750° F estimated for engines in the near future, ref. 3) restricts the choice of bearing steels. Some of the steels being considered for high-temperature bearing applications are the molybdenum tool steels. SAE 52100 modified by the addition of 1 percent aluminum is considered for intermediate temperatures (475° to 600° F, ref. 3). Although very little is known about the rolling-contact fatigue strength of these steels, some of them are being used in aircraft-engine bearing applications on the basis of satisfactory hot hardness and dimensional stability. Laboratory fatigue data on these and other promising high-temperature bearing steels are being compiled. The criterion that a bearing steel must have a satisfactory rolling-contact fatigue life is not readily fulfilled because of the complexity of the fatigue phenomena.

Material fatigue is known to be statistical in nature (ref. 5). Because of the wide scatter in fatigue data, large groups of specimens need to be tested under identical conditions of stress and environment to obtain conclusive results. In the case of bearings, a group (about 30) is tested under fixed conditions of load, speed, and lubrication. As a result of testing thousands of bearings by the bearing manufacturers, a standard definition of life has been developed (ref. 6): the life in millions of inner-race revolutions that 90 percent of the bearings of a given type will exceed under constant test conditions. This definition of life is not stringent enough, according to the authors of reference 4, who feel that, because of the need for high reliability in aircraft engines, the time to earliest failure should be used as a defining criterion.

To improve standard life and eliminate early failures for any bearing steel will require an extensive investigation into the basic mechanism of rolling-contact fatigue. Obtaining fatigue lives by running complete bearings is time-consuming and costly; therefore, it would be advantageous to use a simpler test. This simpler test might have fewer and less complex variables. Styri (ref. 7), in his investigation of simple fatigue tests, was unable to find a test that duplicated both the stress-life relation and the stress pattern for rolling-contact stresses. For bearings the stress-life relation has been determined empirically (ref. 6) as follows: The life of a rolling bearing varies inversely as the ninth to tenth power of the stress. The results as found by Styri are not generally accepted by the industry as being reliable for predicting bearing fatigue lives, because they do not produce in the specimen the same type of stresses that bearing parts experience in operation.

Several bench tests that produce rolling-contact stresses in simplified specimens have been developed. Two examples of these bench tests are the rolling-contact fatigue spin rig (ref. 8) and the rig described in reference 4. These tests produce the same type stress pattern as produced in full-scale bearings, and they accelerate failure time and introduce fewer variables. For the metallographer the restricted location and small size of the stressed volume are a great advantage in studying the effect of various metallurgical factors on rolling-contact fatigue life. To be accepted as reliable, these bench rigs must produce the same kind of failures as found in complete bearings and result in the same stress-life relation. The primary objective of this investigation is, therefore, to establish the validity of the rolling-contact fatigue spin rig.

While final acceptance of a bearing steel must be determined by field tests, these bench tests, if proved acceptable, may be used to isolate the various factors that are detrimental to rolling-contact fatigue so that they may be corrected in bearing manufacture. Reference 9 reports a detailed metallographic investigation of ball failures obtained in the fatigue spin rig, which supports the validity of the rig by showing that the failures are of the same type as experienced in full bearings. Some of the metallurgical factors involved in rolling-contact fatigue and the relative importance of these factors are also discussed. One of the macroscopic factors in rolling-contact fatigue is investigated in reference 10. A study of the effect of fiber orientation on fatigue life and position of failure showed that the nonpolar area was stronger in fatigue than the polar areas under the conditions of the test.

The present investigation was conducted to determine the validity of this bench rig as shown by stress-life exponent and the failure type. Tests were conducted at room temperature with 9/16-inch SAE 52100 balls and cylinders of two tool steels lubricated with a mineral oil. Maximum Hertz stresses of 600,000, 675,000, and 750,000 psi were used.

APPARATUS

Test Rig

Figure 1(a) is a cutaway view of the rolling-contact fatigue spin rig, which is described in detail in reference 8. The test specimens are the two test balls revolving in a plane on the bore of the test cylinder (fig. 1(b)). Air at pressures up to 100 pounds per square inch is introduced through the nozzles to drive the balls at high orbital speeds. The nozzle system and the cylinder are held together by upper and lower cover plates fastened by three removable bolts. The rig assembly is supported from a rigid frame by three flexible cables. In order to keep external constraint at a low value, the drive air is introduced into the rig through a 6-foot-long flexible metal hose.

Operation. - The two test balls separate and maintain relative positions 180° apart at orbital speeds above 150 rpm. Under stable operation the balls revolve in the plane of the nozzle system. Rig operation is analyzed in detail in reference 8.

Loading. - The loading on the balls, which is produced by centrifugal force, is in excess of 700 pounds for a 0.5-inch ball revolving in a 3.25-inch-bore test cylinder at an orbital speed of 30,000 rpm. This will develop a maximum Hertz stress of approximately 750,000 pounds per square inch in the center of the contact ellipse.

Lubrication. - Lubrication is accomplished by introducing droplets of the lubricant into the drive airstream between the guide plates (fig. 1). The rotating airstream forces the droplets against the bore of the test cylinder. Lubricant flow rate is controlled by regulating the pressure at the upstream end of a long capillary tube. The pressure drop through the capillary is sufficient to give excellent control of the flow for small flow rates. The lubricant flow rate used in these tests was approximately 5 milliliters per hour. To make the conditions of the test as nearly comparable as possible with the rating tests given full bearings, a mineral oil (SAE 10) was used; this oil has viscosities of 44 centistokes at 100° F and 6.4 centistokes at 210° F.

Instrumentation. - The two major instrumentation systems of the rig are a speed measurement and control system and a failure detection and shutdown system. Orbital speed is measured by counting the pulses from a photoamplifier on an electronic tachometer. The pulses are generated when the two test balls interrupt a light beam to the photocell. Speed control is accomplished by either manually regulating driving pressure or automatically adjusting pressure to maintain a given speed. When properly adjusted, this system controls the speed to ±120 rpm at 30,000-rpm orbital speed. All the tests were conducted without automatic speed control, because failure of the light source (by burnout or blocking by accumulated oil) would cause the loss of a test; however, a magnetic pulse generator for measuring speed is under development. The test method used is described in the PROCEDURE section.

Failures are detected by comparing the amplified signal from a velocity-type vibration pickup (attached to the rig) with a predetermined signal level that is preset on a meter relay. The large vibration amplitude resulting from a ball or cylinder failure trips the meter relay and shuts down the test rig.

Air supply. - The drive air is dried to less than 30 percent relative humidity and then filtered before being used in the rigs.

Materials. - To serve as a reference base for comparison with bearing data, commercial-melt SAE 52100 was chosen for the ball material.

Cylinder materials were AISI M-1 and AISI M-50 (MV-1) tool steels. Table I gives the nominal analysis of these steels. (Excessive rolling resistance experienced in using SAE 52100 balls on SAE 52100 cylinders prevented the attainment of sufficient speed to reach the desired stress levels; consequently, SAE 52100 cylinders were not used in these tests.)

Inclusion ratings are not available for the 52100 balls and M-1 cylinder heats but were obtained for the cylinder heat of MV-1 steel. Four metallographic specimens from each end and from the center of a 15-inch-long tube forging showed this heat to be exceptionally clean, with a maximum rating of 1.5 in the thin series for all types of inclusions.

Test Specimens

Cylinders. - The nominal dimensions of the test cylinders are as follows: 4.750 inches outside diameter by 3.00 inches long, with an initial nominal inside diameter of 3.250 inches. The bore surface finish was 2 to 3 microinches for all cylinders. Roundness of the bore was held to 0.0001 inch and bore taper to 0.0003 inch maximum. Hardness measurements were taken on the cylinder ends. Each cylinder was uniform within 2 hardness numbers, although the average hardness for different cylinders varied between Rockwell C-60 and C-64.

A cylinder may be used for many tests by regrinding the bore (after approx. 15 tests) 0.060 inch larger and refinishing the surface. This new surface is about 0.022 inch below the location of the maximum-shear-stress location of the previous tests; therefore, the effects of prior stressing are considered to be negligible. Failure positions on one cylinder surface did not correlate with failure positions of the previous test surface.

Balls. - The balls were "off-the-shelf" 9/16-inch-nominal-diameter grade 1 steel balls. Each ball had a fixed track oriented at random to the forging direction of the ball. Two parallel flats 3/16 inch in diameter were ground on each ball, fixing the axis of rotation and allowing the ball to be restarted on the same test track.

PROCEDURE

Pretest Inspection

Cylinders. - Cylinders were inspected for dimension, surface finish, and hardness. This was followed by a magnetic particle inspection for cracks and large subsurface inclusions and a visual inspection for deep scratches and other mechanical damage. Permanent records were kept of all inspection data for correlation with cylinder failures.

Balls. - The test balls were weighed to the nearest milligram, and the test track area was given a ×25 surface inspection. A record was kept of visible defects such as cracks, inclusions, pitting, or mechanical damage that might initiate a failure. Visual inspection was used because nondestructive methods for subsurface inspection such as magnetic particle and ultrasonic testing were not sensitive enough to be useful.

Cleaning and storage. - Before weighing and pretest inspection the balls were cleaned with ethyl alcohol and wiped dry with clean cheesecloth. These balls were stored in water-free varsol for short periods but were coated with a rust preventative for longer periods of storage.

Assembly of Rig

A pair of test balls was assembled in the rig as shown in figure 1(a). Care was taken during assembly to avoid scratching the bore surface. The bore surface and test balls were coated with the test lubricant in order to avoid a dry surface until the lubrication system took effect. Axial position of the test in the cylinder was set and the collet holding the nozzle system tightened. After the rig was installed in the support system, the air and oil lines were connected and the level of the rig checked. Lastly, the light-photocell system was alined and the vibration pickup attached.

Starting and Running Procedure

The rig system was brought up to speed by manually opening the control valve rapidly and smoothly. Close monitoring of the rig speed was necessary because of a gradual decrease in rolling resistance during run-in. This run-in is rapid for the first 2 hours and is practically complete after 48 hours. Speed and oil flow were monitored regularly; and speed, vibration level, and running air pressure were recorded at regular intervals.

The tests were continued uninterrupted for the most part and ran day and night until a ball or race failure actuated the meter relay to shut down the system. All tests were run at ambient temperature (80° F) with SAE 10 mineral oil as the lubricant. Using an average ball weight of 11.827 grams, the load and stress were calculated as outlined in the appendix. Table II summarizes the applied stress for the different balls and cylinders.

Initially, each set of balls was run at a given stress level for 24 hours or until failure. This was to be repeated until 20 to 30 percent of the sample had failed at each stress level. After several tests of 24-hour duration, the absence of failures indicated that the lives would be greater than expected. Therefore, the balls at the 750,000-psi stress level were run to approximately 600 million stress cycles on the ball or until failure, those at the 675,000-psi stress level for 1 billion stress cycles, and those at the 600,000-psi stress level for 1.4 billion stress cycles. When one ball of a test pair failed, the surviving ball was paired with another surviving ball of the same test group and tested to the runout point. In the case of failure of the cylinder, the test balls were rerun unless there was mechanical damage to the balls. Several tests were run on each cylinder track until it failed.

RESULTS AND DISCUSSION

Stress-Life Relation

The failures in each group of balls are plotted on Weibull paper (appendix) in order to interpolate the 10-percent life. These are shown in figure 2 with calculated best-fit lines. Figure 3 is a stress-life plot for all the ball data. The line drawn is the logarithm of the stress-life equation calculated by the method of least squares using the interpolated 10-percent lives.

In analyzing cylinder-life data, all unfailed tracks whose running time did not exceed 50 percent of the maximum failure life are not included in the sample in order to give a conservative value for the 10-percent life for the cylinders. Data for the MV-1 cylinders at the 600,000- and 750,000-psi stress levels are plotted on Weibull paper in figure 4. These data are summarized in figure 5.

Balls. - After testing 50 balls at each stress and plotting the failures on Weibull paper, the failure distribution for the 750,000-psi stress level appeared to fall into two groups. In order to determine whether this grouping was real or whether it resulted from a statistically insufficient sample, an additional 24 balls from the same heat were tested at this stress. The resulting plot (fig. 2(c)) shows that the distribution followed the characteristic pattern for bearing failures. In view of the long testing times involved, no additional data were taken for the other two stress levels.

The slope of the line of figure 3 is the stress-life exponent; that is, the life varied inversely as the 10.4 power of the applied stress. This is in reasonably good agreement with the accepted value of 9 to 10 given in reference 6. The slopes of the best-fit lines in figures 2(a) and (c) are approximately the same. With the use of the 10-percent lives from these two figures, the calculated stress-life exponent is equal to approximately 10. The data for the 675,000-psi stress level only shift the line without changing the exponent appreciably. The reason for the longer than expected lives for the 675,000-psi stress level is not apparent and must be looked for in a metallographic examination of these balls. The reliability of the 10-percent lives will improve with increased sample size.

Cylinders. - Failure data on the cylinders were obtained as a by-product of the test. Because these tool steels are inferior in fatigue to SAE 52100, a number of cylinder failures occurred during this investigation. Four heats of cylinder material were used, two of commercial-melt AISI M-1 and one each of AISI M-50 (MV-1) and vacuum-melt AISI M-1. Each cylinder material was tested at two stress levels. The stress-life exponents for the cylinders are not too well established, because data at only two stress levels were obtained. However, in view of the agreement of the ball stress-life exponent with the accepted value, the cylinder data are thought to be of interest. Figure 5 summarizes the data for the AISI M-50 (MV-1) steel. The calculated exponent is 9.7.

The results for the AISI M-1 steel are anomalous. Lives of 12.8×10^6 cycles for 750,000-psi stress level and 19.0×10^6 cycles for the 675,000-psi stress level were obtained. It is believed that an insufficient number of failures was responsible for this result. Other possible reasons could have been an insufficient separation of the stress levels or a defective heat of material. The stress-life exponent for AISI M-50 (MV-1) steel is in good agreement with the accepted value. The MV-1 material also appears to have better rolling-contact fatigue life than the AISI M-1 steel for the heats of material used in these tests. An insufficient number of failures for a heat of vacuum-melted AISI M-1 cylinders did not allow a comparison to be made with the commercial-melt AISI M-1.

Failure Type

The fatigue failures obtained in the spin rig are of the same general type as observed in full bearings. They are both characterized as being localized in nature and limited in depth of spalling and are usually of subsurface origin near the plane of maximum shear stress. More detail was observed in ball failures from this rig than in a bearing failure, because failure can be stopped at an earlier stage.

Figure 6 includes representative photographs of a bearing race failure (tested conventionally) and ball failures (tested in the spin rig). The section views were taken parallel to the plane of the contact track. Macrophotographs of a failed bearing race (size 222) and a ball failure obtained in the spin rig are shown in figures 6(a) and (d), respectively. The failures tend to cover the full width of the track.

The section views of figures 6(b) and (e) show that the spalling tends to be shallow in extent. Propagation cracks may be seen in both sections. The small inset (fig. 6(g)) illustrates the amount of detail that may be obtained from an early stage of ball test failure. The views of figures 6(c) and (f) show typical inclusion damage in the form of incipient cracking of the matrix. Such inclusion damage is more pronounced (in the number of inclusions affected and the extent of cracking) in the ball tests because of the higher stress concentration. Subsurface origin of failures has been observed for bearings (ref. 7) and was observed for ball failures in reference 9.

With regard to similarity of the stress-life relation and similarity of failure produced between the spin rig and full-sized bearings, the spin rig is an acceptable test for comparing materials for use as either bearing balls or races.

SUMMARY OF RESULTS

Groups of 9/16-inch SAE 52100 grade 1 steel balls were tested at 600,000-, 675,000-, and 750,000-psi stress levels under room ambient temperature conditions using SAE 10 mineral oil as lubricant. Cylinders of both vacuum- and commercial-melt AISI M-1 and commercial-melt AISI M-50 (MV-1) steels with bore sizes from 3.250 to 3.550 inches were used as the race specimens. The results of these tests are as follows:

The stress-life exponents for the balls (10.4) and MV-1 cylinders (9.7) are in good agreement with the accepted value (9 to 10) obtained in full bearing tests. Anomalous results were obtained from a group of AISI M-1 cylinders. Fatigue spalls of subsurface origin similar to failures obtained in bearings were obtained. The comparable failure type and stress-life exponents obtained indicate that this bench test is an acceptable method for comparing materials for use as bearing races and balls as well as for investigating other factors in rolling-contact fatigue.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, December 12, 1956

APPENDIX - STRESS CALCULATIONS AND PRESENTATION OF BEARING FATIGUE DATA

Stress calculations. - The load can be calculated from the speed, bore diameter, and ball weight:

$$P = AWrN^2 \text{ pounds}$$

where W is the ball weight in grams, r is the pitch radius of the ball set in inches, N is the system rpm, and A is a dimensional constant.

If the load and the principal curvatures of the specimens are known, the stress may be calculated from the modified Hertz equations given in reference 11.

Calculation of stress-life exponent. - The general equation for bearing fatigue life is

$$\frac{L_1}{L_2} = \left(\frac{S_2}{S_1}\right)^x$$

Then

$$x = \frac{\log \frac{L_1}{L_2}}{\log \frac{S_2}{S_1}}$$

where L_1 and L_2 are 10-percent lives at stresses S_1 and S_2.

Presentation of bearing fatigue data. - A distribution function developed by Weibull has been found to fit the observed scatter in the fatigue lives of rolling bearings (ref. 12). Because of the usually small sample size involved (about 30), the data cannot reliably be fitted into a frequency curve. Instead, the cumulative form of the distribution is used. The cumulative distribution function (Weibull) is as follows:

$$\log \frac{1}{S(L)} = KL^e \tag{1}$$

where $S(L)$ is the statistical percent of the sample surviving the first L stress cycles, and K and e are positive constants.

Data are plotted on special probability paper on which the Weibull distribution becomes a straight line of slope e. The ordinate represents log-log $1/S(L)$, but is graduated in terms of the statistical percent failed at L stress cycles.

A set of data is ordered according to life, and each succeeding life is given a rank (statistical percent). If the median rank is used, a best-fit line can be drawn in by eye which takes the general direction of the array of points and which splits the array in half. A median rank is an estimate of the true rank in the population that has an equal probability of being too large or too small. The best-fit lines (Weibull plot) for this report were calculated because of the limited number of failures in each test group.

REFERENCES

1. NACA Subcommittee on Lubrication and Wear: Review of Current and Anticipated Lubricant Problems in Turbojet Engines. NACA RM 51D20, 1951.

2. Perlmutter, Isaac: Rolling Contact Bearings for High Temperature Applications. Memo. Rep. WCRTL-M-5620, Materials Lab., Res. Div., Wright-Patterson Air Force Base, Dayton (Ohio), Apr. 15, 1952.

3. SAE Panel on High-Speed Rolling-Contact Bearings: Trends of Rolling-Contact Bearings as Applied to Aircraft Gas-Turbine Engines. NACA TN 3110, 1954.

4. Barnes, Gilbert C., and Ryder, Earle A.: A Look at Some Turbine Bearing Problems. Preprint No. 693, SAE, 1956.

5. Epremian, E., and Mehl, R. F.: Investigation of Statistical Nature of Fatigue Properties. NACA TN 2719, 1952.

6. Lundberg, G., and Palmgren, A.: Dynamic Capacity of Rolling Bearings. ACTA Polytech., Mech. Eng. Ser., vol. 1, no. 3, 1947.

7. Styri, Haakon: Fatigue Strength of Ball Bearing Races and Heat-Treated 52100 Steel Specimens. Proc. A.S.T.M., vol. 51, 1951, pp. 682-700.

8. Macks, E. F.: The Fatigue Spin Rig - A New Apparatus for Rapidly Evaluating Materials and Lubricants for Rolling Contact. Lubrication Eng., vol. 9, no. 5, Oct. 1953, pp. 254-258.

9. Bear, H. Robert, and Butler, Robert H.: Preliminary Metallographic Studies of Ball Fatigue Under Rolling-Contact Conditions. NACA TN 3925, 1957.

10. Butler, Robert H., Bear, H. Robert, and Carter, Thomas L.: Effect of Fiber Orientation on Ball Failures Under Rolling-Contact Conditions. NACA TN 3933, 1957.

11. Jones, A. B.: New Departure - Analysis of Stresses and Deflections. Vols. 1 and 2. New Departure, Div. General Motors Corp., Bristol (Conn.), 1946.

12. Johnson, Leonard G.: The Median Ranks of Sample Values in Their Population with an Application to Certain Fatigue Studies. Ind. Math., vol. 2, 1951, pp. 1-9.

TABLE I. - NOMINAL COMPOSITION OF TEST STEELS

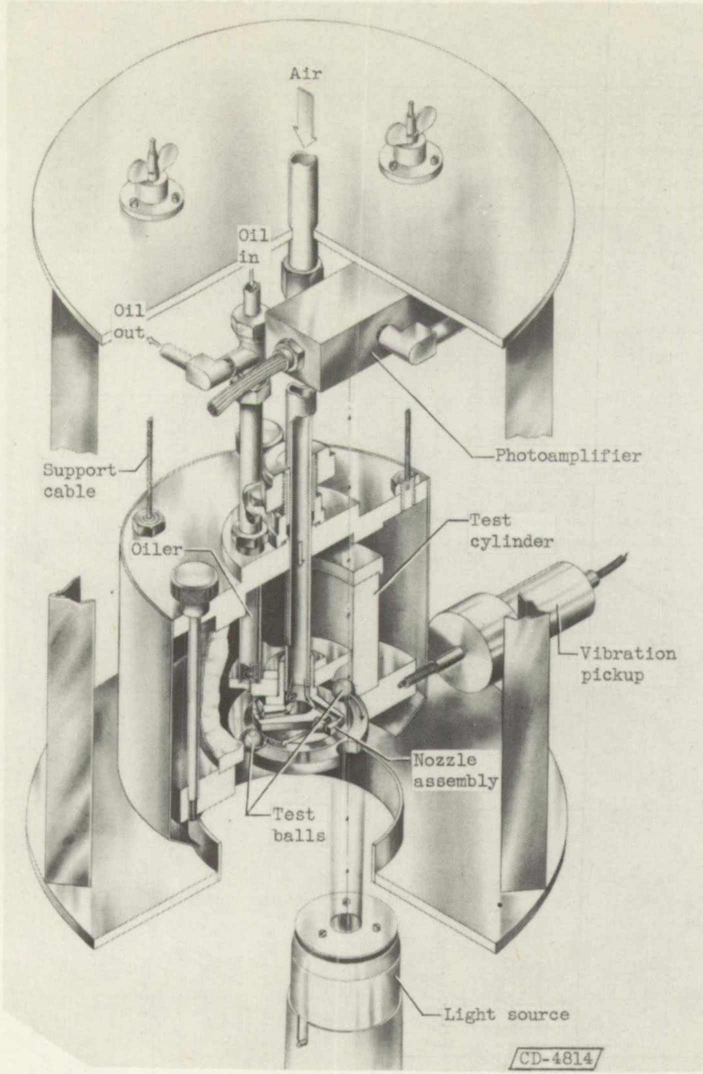
Steel	Nominal analysis, percent								
	C	Si	Mn	P (max.)	S (max.)	Cr	W	Mo	V
SAE 52100[a]	0.95-1.10	0.20-0.35	0.25-0.45	0.025	0.025	1.30-1.60	----	----	----
AISI M-1[b]	0.80	0.30	0.25	-----	-----	4.00	1.50	8.50	1.00
AISI M-50 (MV-1)[b]	0.80	0.25	0.30	-----	-----	4.10	----	4.25	1.10

[a]SAE Handbook, 1947, p. 286; balls only.

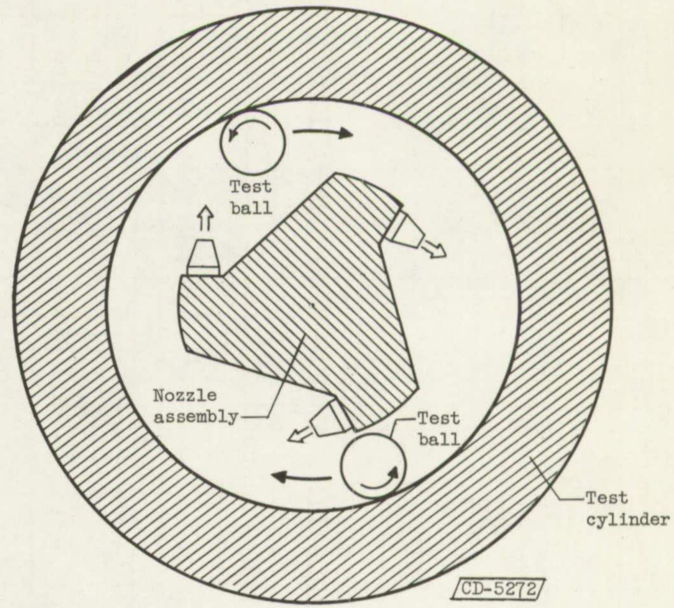
[b]Manufacturer's data; cylinders only.

TABLE II. - TEST STRESS FOR VARIOUS SPECIMENS

Material	Applied stress, psi	Sample size	Ball diam., in.	Bore diam., in.
SAE 52100	750,000 675,000 600,000	74 balls 50 balls 50 balls	9/16 9/16 9/16	------ ------ ------
MV-1 (AISI M-50)	750,000 600,000	1 cylinder 2 cylinders	---- ----	3.250 3.250
AISI M-1	750,000 675,000	2 cylinders 4 cylinders	---- ----	3.550 3.430 3.490 3.370 3.370 3.430
AISI M-1 (Vacuum-melted)	750,000 675,000 600,000	1 cylinder 1 cylinder 2 cylinders	---- ---- ----	3.310 3.310 3.310 3.310

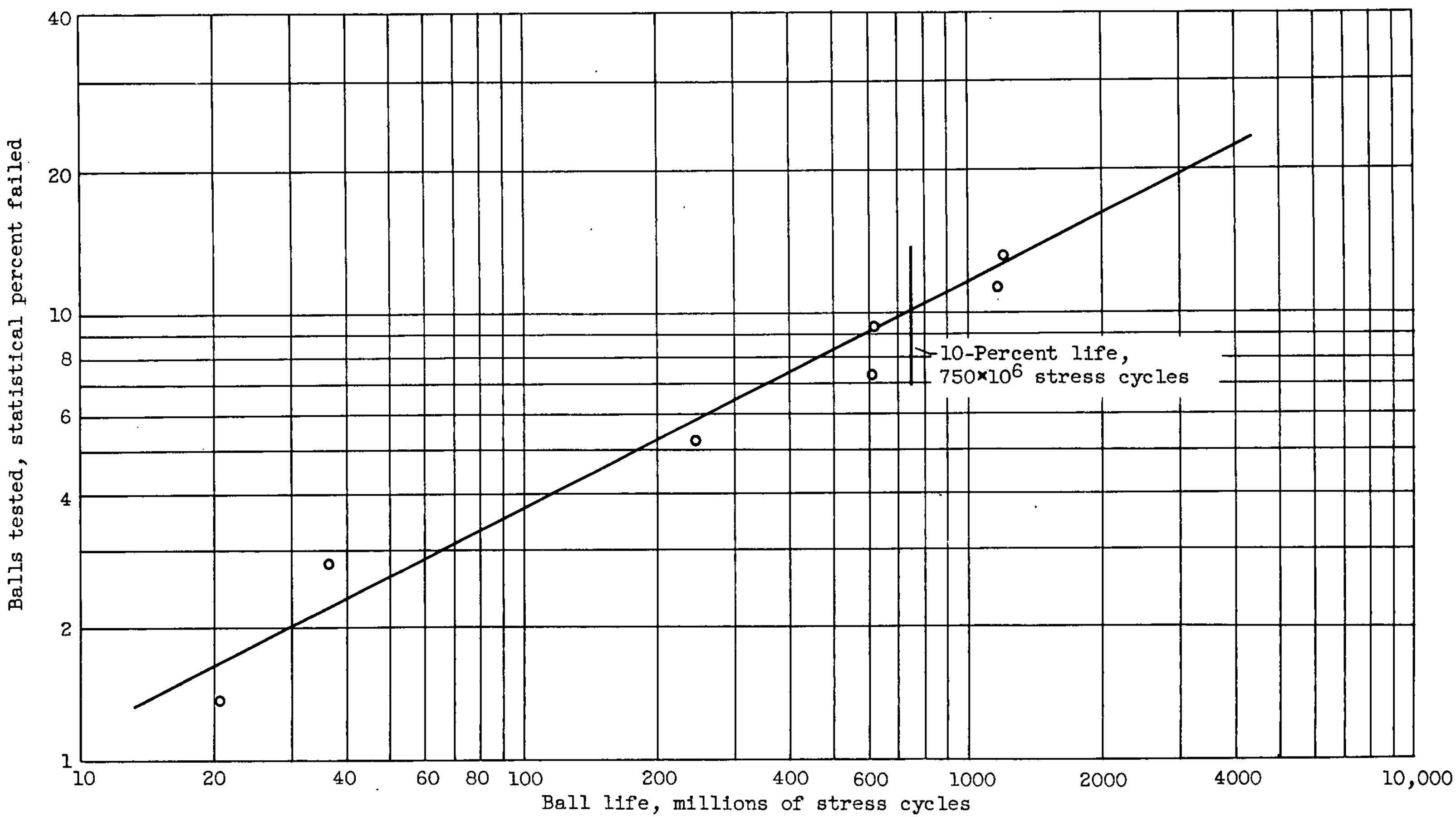


(a) Cutaway view.



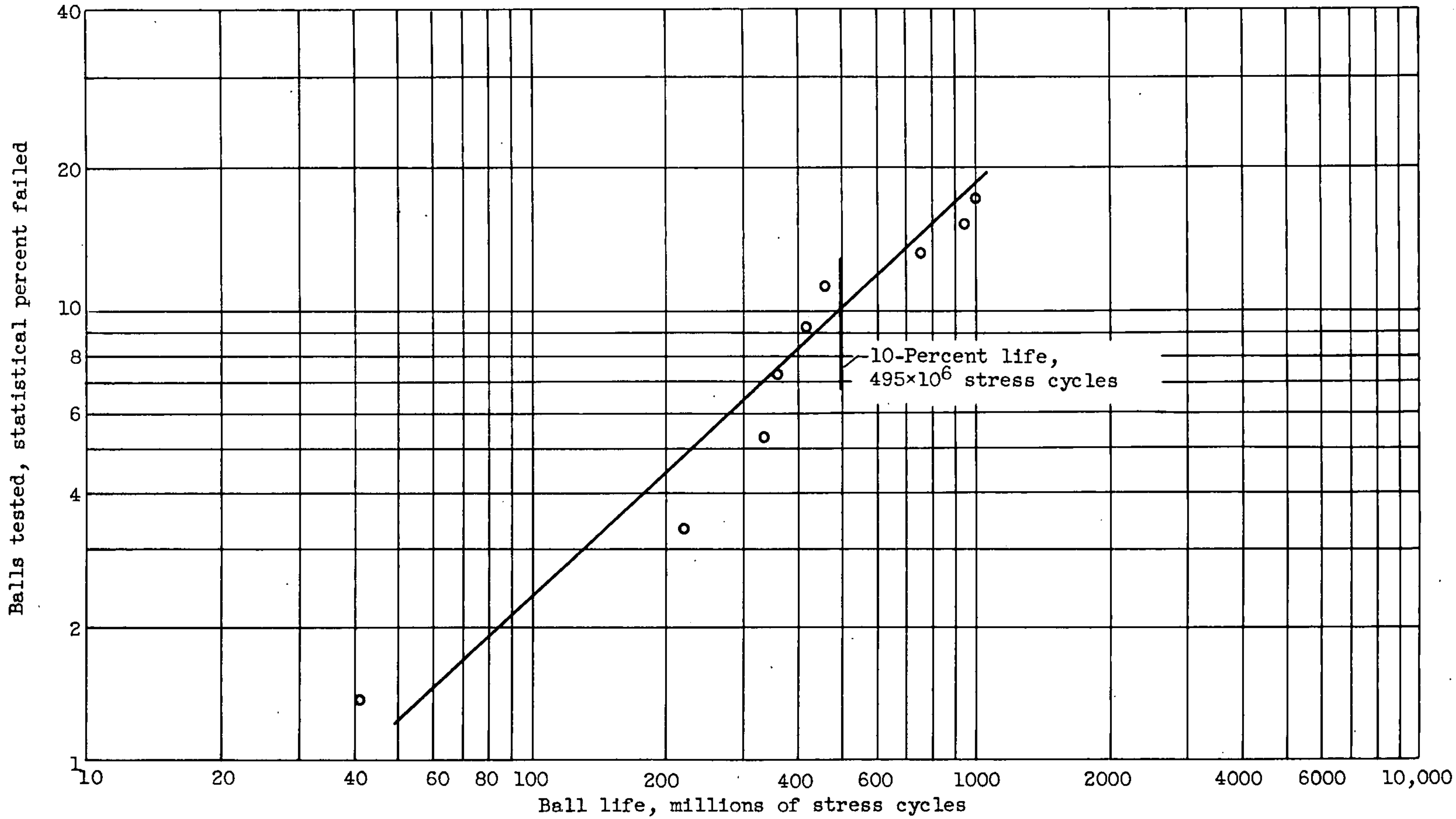
(b) Schematic diagram.

Figure 1. - Rolling-contact fatigue spin rig.



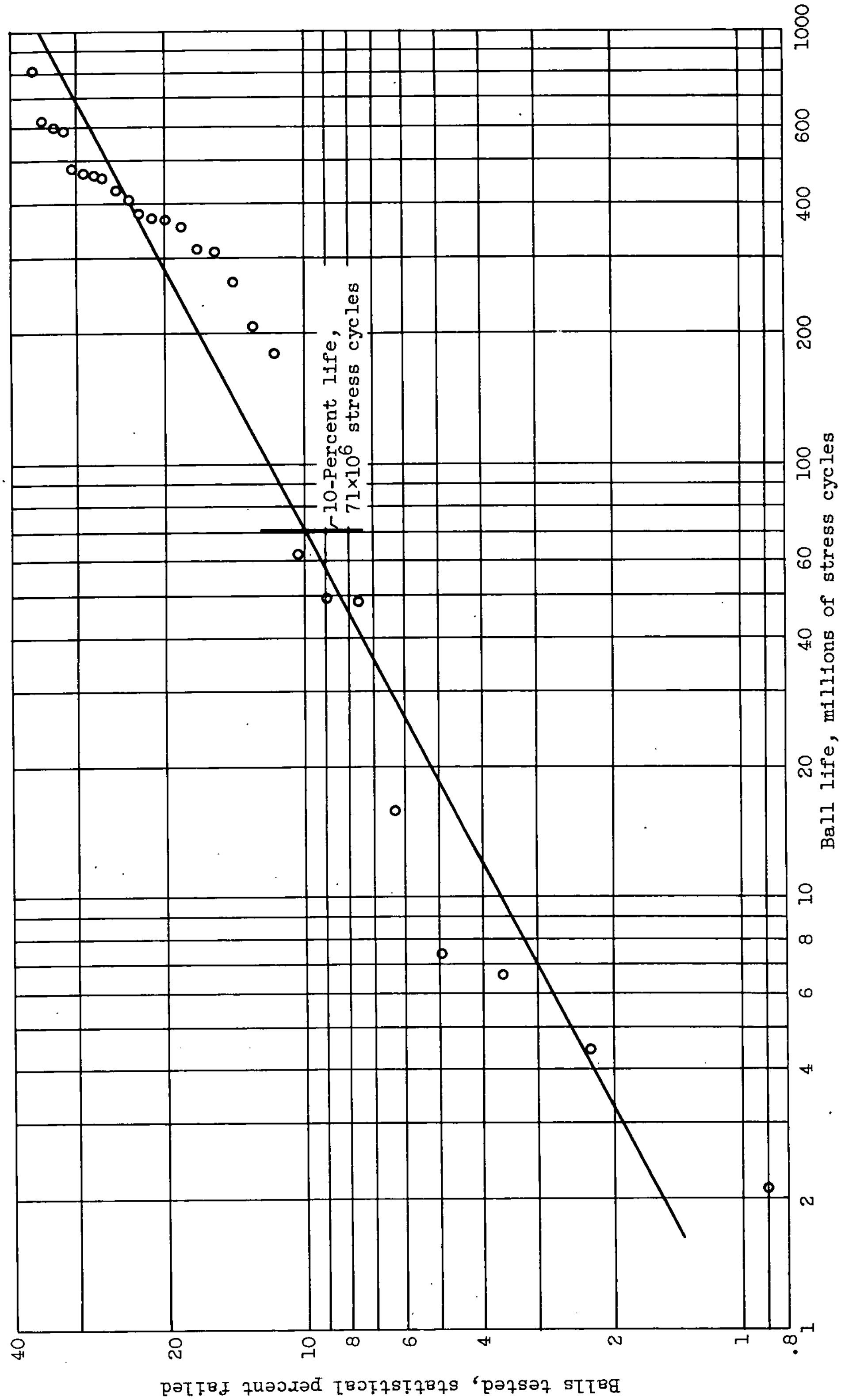
(a) Maximum Hertz stress, 600,000 psi; sample, 50 balls; 7 failures.

Figure 2. - Life of 9/16-inch SAE 52100 grade 1 balls at room temperature. Lubrication, SAE 10 mineral oil at 5 milliliters per hour.



(b) Maximum Hertz stress, 675,000 psi; sample, 50 balls; 9 failures.

Figure 2. - Continued. Life of 9/16-inch SAE 52100 grade 1 balls at room temperature. Lubrication, SAE 10 mineral oil at 5 milliliters per hour.



(c) Maximum Hertz stress, 750,000 psi; sample, 74 balls; 27 failures.

Figure 2. - Concluded. Life of 9/16-inch SAE 52100 grade 1 balls at room temperature. Lubrication, SAE 10 mineral oil at 5 milliliters per hour.

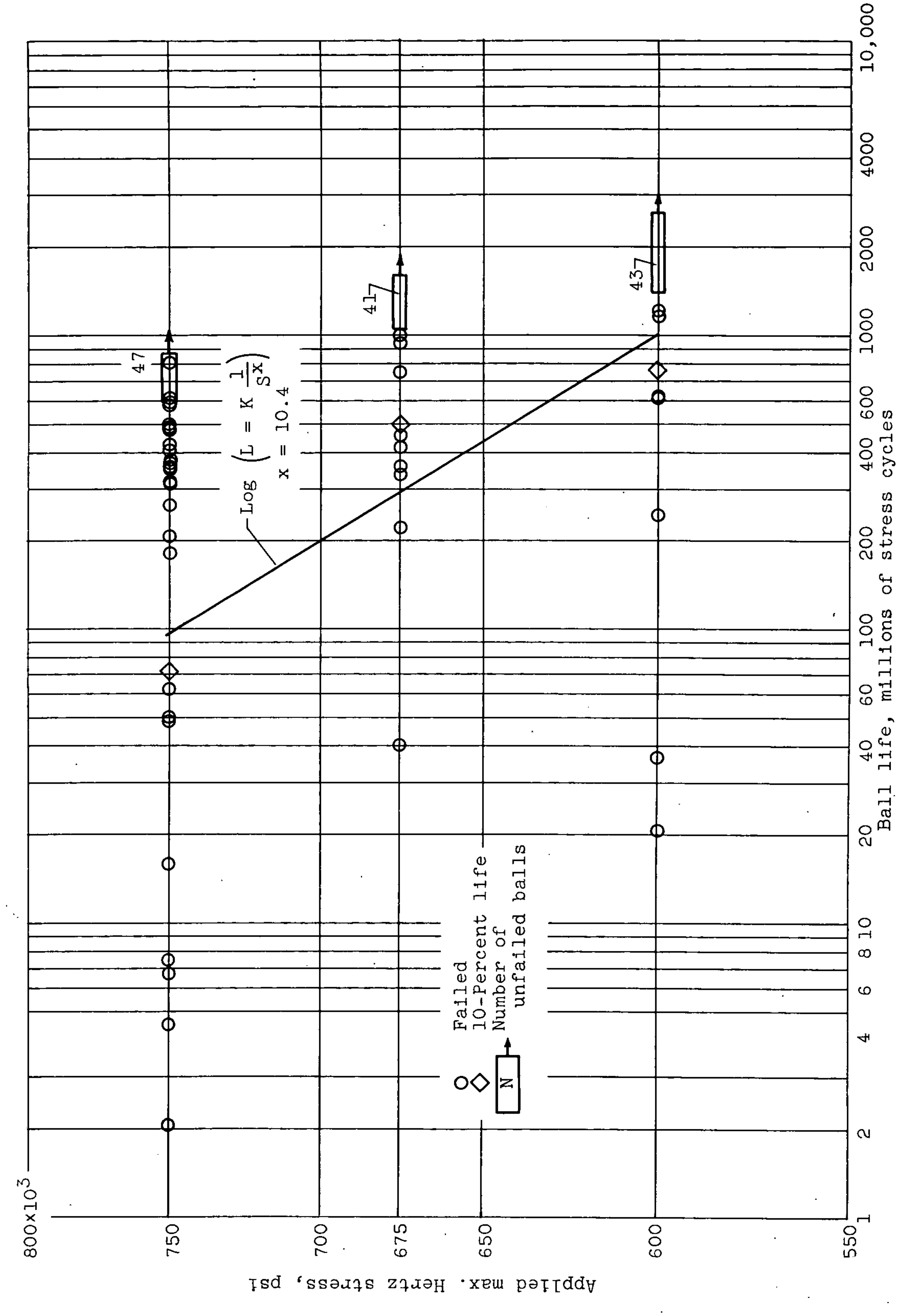
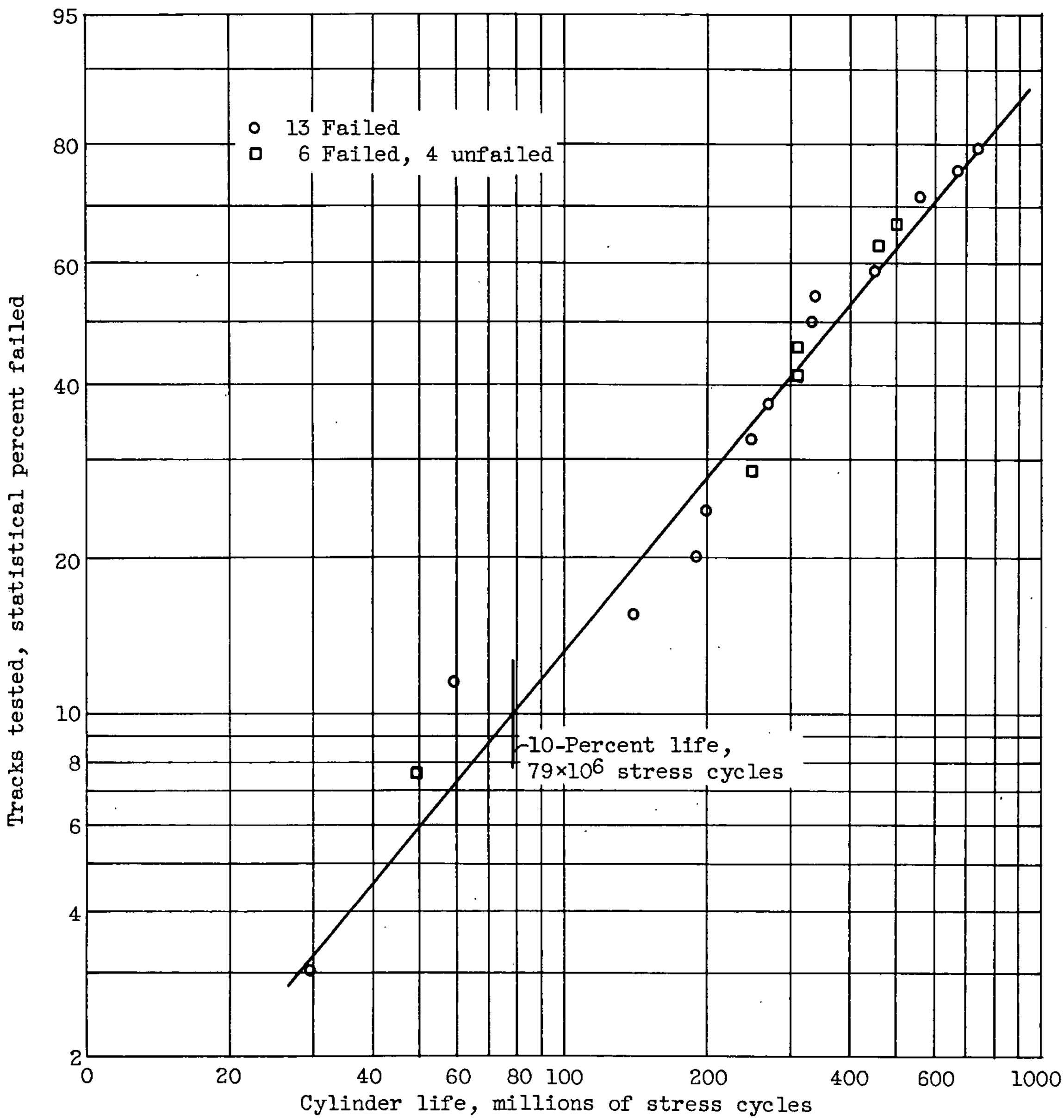
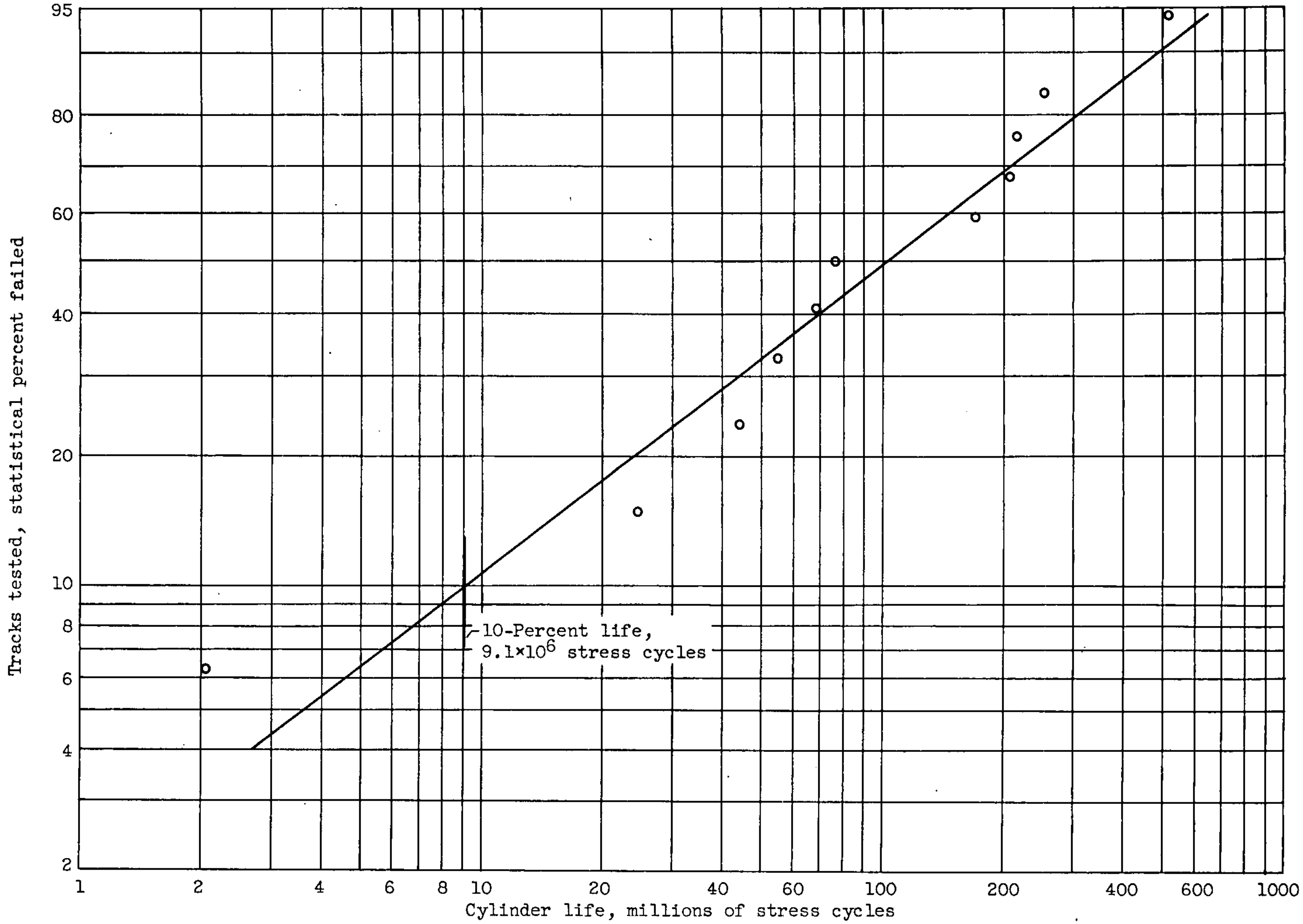


Figure 3. - Variation of ball life with applied stress. Balls, grade 1, 9/16-inch SAE 52100 steel; room temperature; lubrication, SAE 10 mineral oil at 5 milliliters per hour.



(a) Maximum Hertz stress, 600,000 psi; sample, 23 tracks (two cylinders); 19 failures.

Figure 4. - Life of MV-1 tool-steel cylinders at room temperature. Lubrication, SAE 10 mineral oil at 5 milliliters per hour.



(b) Maximum Hertz stress, 750,000 psi; sample, 11 tracks; 11 failures.

Figure 4. - Concluded. Life of MV-1 tool-steel cylinders at room temperature. Lubrication, SAE 10 mineral oil at 5 milliliters per hour.

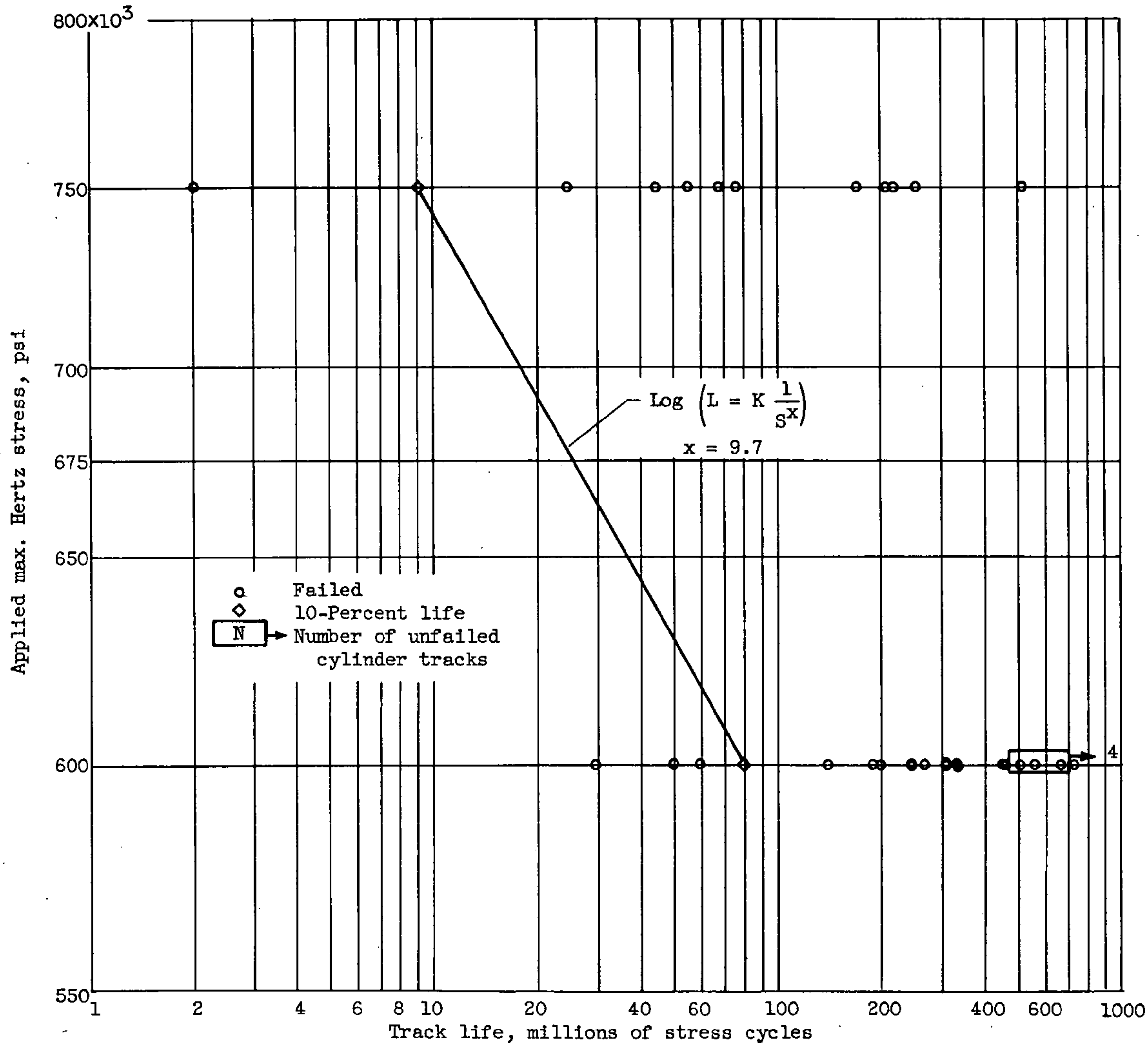
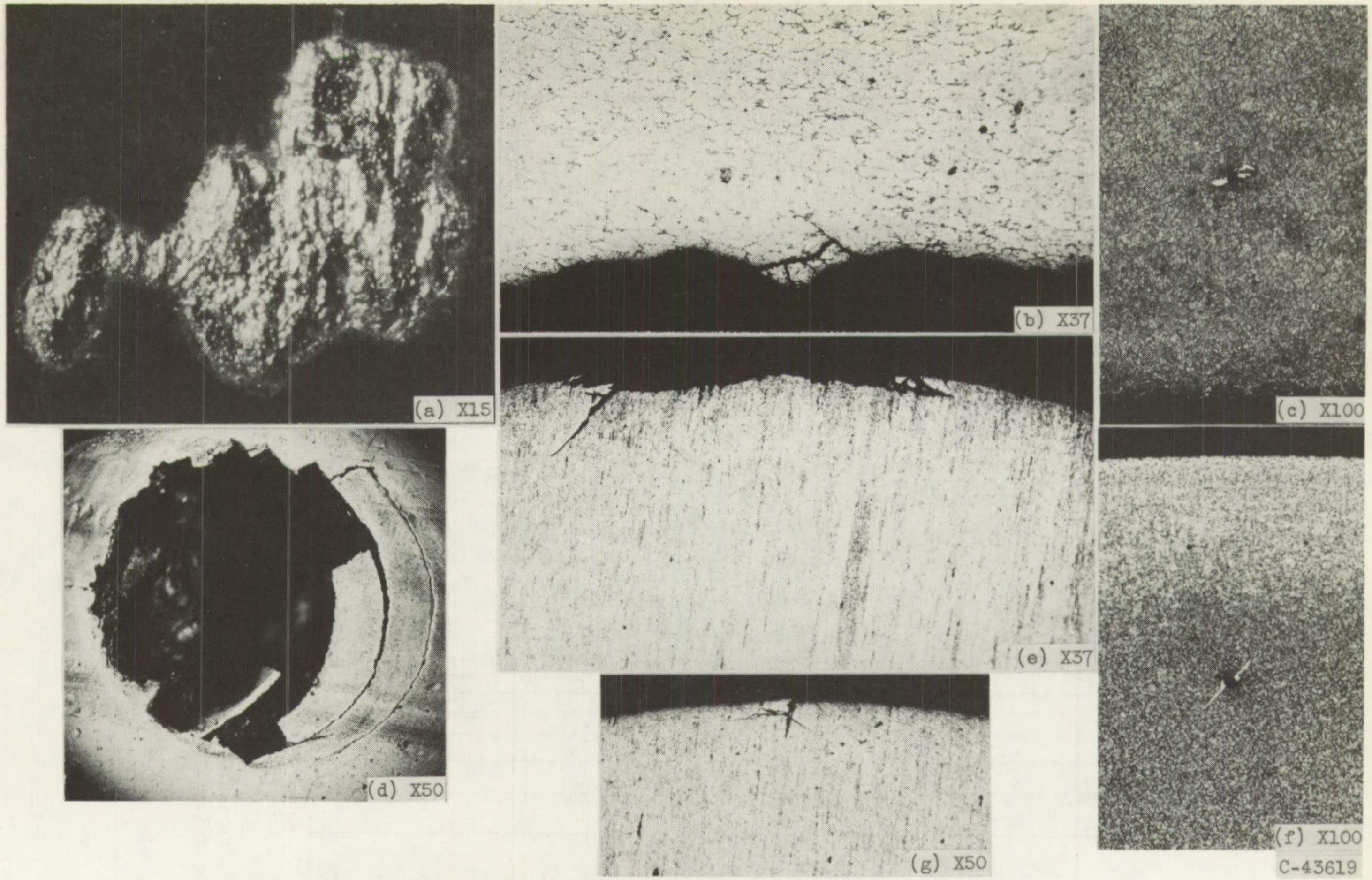


Figure 5. - Variation of track life with applied stress. Cylinders, MV-1 tool steel; room temperature; lubrication, SAE 10 mineral oil at 5 milliliters per hour.



(a) Small spall on inner race of M-1 (222) bearing.
(b) Section view of part of same spall.
(c) Incipient failure near point of maximum shear stress.
(d) Typical spall, spin-rig-tested ball.
(e) Section view of spall.
(f) Incipient failure near point of maximum shear stress.
(g) Section view of early failure on spin-rig-tested ball.

Figure 6. - Comparison of failures of a bearing inner race and balls tested in rolling-contact fatigue spin-rig.